Performance of Solar-Powered Vapor-Jet Refrigeration Systems with Selected Working Fluids

V. W. Chai and F. L. Lansing
DSN Engineering Section

The performance of the solar-powered vapor-jet refrigeration scheme is compared with five selected working fluids: R-11, R-12, R-113, Butane and Water. These fluids were selected among those able to suit both a power cycle and a refrigeration cycle. The results indicated that water has the highest coefficient of performance and differs by a wide margin compared to the other compound organic fluids at all boiler and evaporator temperatures considered.

I. Introduction

One of the objectives of the on-going DSN Energy Conservation Project at the Deep Space Communication Complex (GDSCC), Goldstone, California, is to reduce the purchased energy by using alternate nondepletable sources such as solar energy. Heating, ventilation, and air-conditioning (HVAC) systems consume about 30 percent of the total energy consumption and their coupling with solar energy presents a potential saving. It has been shown earlier (Ref. 1) that the solar-powered vapor-jet refrigeration system (Fig. 1) possesses an overall coefficient of performance comparable with that obtained from solar-absorption systems or solar-powered Rankine turbo-compressor systems. However, little has been done in the literature to explore the potential use of solar-jet refrigeration systems, and it is the intent of this article to help shed some light on the subject. Conceptually, the solar-jet refrigeration scheme as sketched in Fig. 1 is similar to solar-powered Rankine turbo-compressor scheme in which a Rankine-power cycle drives a vapor-compression refrigeration cycle. The former scheme uses a nozzle-diffuser ejector for the expansion-compression processes instead of the combined turbine compressor unit used in the latter scheme.

Another important difference between the two schemes is that the ejector scheme can only operate with a single type fluid while the turbine-compressor scheme can work with dual fluid loops. Consequently, it is very important to carefully select the working fluid which best obtains not only a high power cycle efficiency, but a high coefficient of performance (COP) for the refrigeration cycle also. Other than water as the working fluid (Ref. 1), Anderson (Ref. 2) used butane gas while Kakabaev and Davletov (Ref. 3) used freon refrigerant 12 in their jet refrigeration studies. Neither stated the rationale behind using the chosen chemical as the working fluid.

In the literature, various working fluids for Rankine power cycle alone or refrigeration cycles alone have been studied extensively (Ref. 4, 5, 7). However, the selection of fluids to suit both a power cycle and a refrigeration cycle has not been explored enough, despite the recent interest in solar Rankine turbo-compressor refrigeration systems.

II. Criteria for Working Fluid Selection

There are literally hundreds of organic compounds that can be used as working fluids. Some are only suitable for power cycles which require chemical stability at high boiler temperature while others are only suitable for refrigeration cycles that operate at very low temperature ranges without freezing. Before selecting the candidate fluids that have properties suitable to both a power cycle and a refrigeration cycle to be used in the solar jet refrigeration system analysis, it is worthwhile to consider the selection criteria. The fluid thermal behavior which maximizes the system coefficient of performance (COP) and satisfies these criteria will be selected. The selection criteria are listed as follows:

First, the availability of the fluid and the associated cost. A fluid which may result in a very high performance, but has limited resources or is scarcely found would mean a high cost for the system.

Second, candidate fluids should have a relatively high critical temperature and pressure which raises the high limit for the boiler temperature. A boiler temperature of 93.3°C (200°F) for example, means that many fluids having low critical temperatures such as refrigerant-13 (T critical = 29°C) would be operating at supercritical conditions. Supercritical operation requires a careful nozzle/diffuser design. The boiler to evaporator pressure ratio will also affect the nozzle design. For example, for superheated steam, if the back pressure is lower than 54.6 percent of the boiler stagnation pressure, the nozzle must be convergent-divergent with supersonic flow. Without a proper nozzle/diffuser design, under-expansion or over-expansion will occur, which decreases the isentropic efficiency.

Third, thermal and chemical stability under cycling between hot and cold regions is another criterion controlling the highest boiler temperature for longer operation.

Fourth, for safe operation candidate fluids should be compatible with surrounding materials such as copper, iron, rubber seals, etc.

Fifth, fluids should not have excessive boiler pressures or too low condensing pressures. For a particular boiler temperature, the saturation pressure should not be too high, to prevent leakage problems, and eliminate the need for heavy components. Also, condenser or evaporator pressures should not be too low in order to avoid air leakage into the system. The condenser to evaporator pressure ratio should be within allowable limits to assist in an efficient diffuser design.

Sixth, the slope of the vapor saturation curve on the temperature-entropy diagram should be as steep as possible. This slope defines the exit conditions of the expansion and compression processes, and consequently affects the flow rates of the system. For isentropic expansion or compression, with fluid having a negative slope, exit conditions will be wet vapor at the end of the expansion process and superheated vapor at the end of the compression process as shown in Fig. 2a. Conversely, for a positive slope vapor saturation curve as shown in Fig. 2b, the opposite conditions take place. It is therefore preferable to use a fluid having a steep slope for its vapor saturation curve. The slope should not be greatly deviated from the vertical to avoid extreme conditions.

Seventh, include other operational safety criteria such toxicity, flammability, and chemical reactions with air or water. Finally, the candidate fluid should have high heat transfer coefficient within the temperature range of operation. This property reduces the heat transfer area required for the solar collector, condenser and evaporator heat exchangers.

III. Selected Working Fluids

Based on the above criteria mentioned, five fluids are chosen for investigation, and their properties are listed in Table 1. These selected fluids are known to be suitable in other solar-powered air-conditioning schemes. Steam jet refrigeration is a well known technology and it has been widely used in industry for some time before the use of mechanically-driven vapor-compression refrigeration. Water has its well known advantages. However, it is limited to temperatures above its freezing temperature. Also, evaporator operation, with water, at low temperature and pressure has resulted in high specific volumes which required bulky equipment. Refrigerant-12 was used later in the freon ejector solar cooler reported by Kakabaev and Davletov (Ref. 3). With a boiler temperature ranging from 70 to 80°C, a condenser temperature at 38°C and an evaporation temperature of 18°C, the coefficient of performance (COP) ranged from 0.3 to 0.7 depending on the cooling load and the mass flow ratio between the high pressure vapor into the nozzle and the low pressure vapor from the evaporator. Accordingly, R-12 is selected in this work as a prospective candidate to be compared with water.

Biancardi, Mender, Blecher and Hall (Ref. 6) used refrigerant-11 in their turbo-compressor unit. A COP of about 1.0 was achieved with turbine inlet temperature of 115.55°C (240°F). Refrigerant-11 resulted in not only a high COP but also proved itself adequate in both power cycle and refrigeration cycle combination. Therefore, R-11 is selected in this study as another candidate fluid to be added to the above list.

Barber (Ref. 4) used both refrigerant-12 and refrigerant-113 for the dual fluid solar-powered Rankine turbocompressor cycle. Refrigerant-113 is often considered as a power cycle fluid (Ref. 5) for its high critical temperature. Refrigerant-113 is then added to the list since it has also a high thermal stability at temperatures up to 500°C.

Finally, after Anderson (Ref. 2), butane is considered a candidate fluid for the study. With butane, the performance was shown in Reference 2 to be comparable with that of the Rankine turbocompressor system and the ejector system can operate at lower supply temperatures than that for solar absorption system.

IV. Performance Comparison With the Selected Working Fluids

The five working fluid candidates namely; water, R-11, R-12, R-113 and Butane are tried each in a jet refrigeration cycle under identical evaporator, condenser and boiler temperatures. The objectives were set to determine the trend of the cycle coefficient of performance with (1) various boiler temperatures keeping all other parameters the same and (2) various evaporator temperatures keeping all other parameters the same.

The isentropic efficiency for each of the nozzle and the diffuser is kept unchanged in the study. Since the condenser temperature can vary only with the type of cooling medium whether it is air or water, and ambient conditions, the condenser temperature was considered a fixed parameter not affecting the final fluid choice. On the other hand, the boiler temperature is considered an important parameter since it is controlled by the maximum stable temperature the fluid can handle. The evaporator temperature is also important since it is controlled by the fluid freezing temperature.

The results of the parametrization study are plotted in Figures 3 and 4 using the fluid thermodynamic properties in Reference 8 and the nomogram methodology previously reported in Reference 1. Figure 3 shows the coefficient of performance (COP) versus the evaporator temperature for the five working fluids with all other parameters fixed. Since water cannot operate below 0°C, the evaporator temperature range was taken between 4.44°C (40°F) and 15.56°C (60°F). From Fig. 3, the higher the evaporator temperature, the higher the COP which is an expected result derived from the ideal Carnot's performance.

It is also evident from Fig. 3 that the COP for fluids R-11, R-12, R-113 and Butane lie within a narrow band much lower than that for water. For example, at 10°C (50°F) evaporator temperature, the COP for the organic fluids tested varied from 0.4 to R-11 to 0.47 for R-113 compared to 0.85 for water at the same conditions: a result which makes water a fluid with excellent performance.

In Fig. 4, the COP results were plotted with fixed evaporator temperature versus the boiler temperature. Boiler temperatures were chosen between 80°C (176°F) and 110°C (230°F) to suit most low cost flat-plate collectors. Theoretically for the Carnot refrigeration cycle, the COP increases with increasing the high source temperature (boiling temperature) and the trends in Fig. 4 for water follow the theoretical trend. However, for refrigerant-11 and refrigerant-113, the COP reaches a maximum at about 95°C with COP at about 0.35 for R-113 and 0.32 for R-12. For refrigerant-12 the COP increases with increasing boiler temperature, but its relatively low critical temperature (~110°C) prevents it from operating at higher temperature values.

Also from Fig. 4, steam has the highest COP among the fluids selected, which is contributed mainly by its large latent heat of vaporization. On the other hand, steam required bulky components to handle the large volumes of flow since it has a very large specific volume compared to organic compounds at the same temperature.

V. Summary and Findings

From this study, the following findings can be drawn:

1. The performance of the solar-powered vapor-jet refrigeration scheme was tested with five selected working fluids; R-11, R-12, R-113, Butane and water, using the nomogram methodology constructed earlier in Reference 1.
2. The proper selection of the working fluid is subject to many selection criteria such as (1) fluid availability and low cost, (2) high critical temperature (3) thermal and chemical stability under hot/cold cycling (4) compatibility with other surrounding materials (5) low saturation pressures, (6) steep slope of vapor-saturation curve on temperature-entropy diagram (7) nonflammability, toxicity properties, and (8) high heat transfer coefficient.
3. All five working fluids selected have an increasing trend of COP with the evaporator temperature. The increasing trend could be reasoned by the similar Carnot's behavior. However, the resulting COP had two distinct bands separated by a large margin; water performance in one side and all the other organic compounds in a narrow band. Water was shown superior than the other organic fluids with a (COP) almost double the value obtained from the other fluids.
4. With the boiler temperature as a parameter, refrigerants R-11 and R-113 reached a maximum (COP) value at about 95°C but water and R-12 had a monotonic trend. Again,

the water (COP) was superior compared to other organic compounds. The large latent heat of vaporization that the water possesses in addition to the odd-shaped and small vapor curve slopes that the organic compounds have on the temperature-entropy diagram contributed to the water performance superiority.

Although water, as a fluid, has the disadvantage of large specific volume at low temperatures which tend to increase the component size, besides its low freezing temperature which requires careful handling and design, it is foreseen that water will be selected as the only candidate in future studies of solar-powered vapor-jet refrigeration schemes.

References

1. Lansing, F. L., Chai, V. W., "A Thermodynamic Analysis of a Solar-powered Jet Refrigeration System". DSN Progress Report 42-41, Jet Propulsion Laboratory, Oct. 15, 1977, pp. 209-217.

2. Anderson, H., "Assessment of Solar Powered Vapor Jet Air-conditioning System". Presented at 1975 International Solar Energy Congress and Exposition (ISES) held at UCLA, Los Angeles, California, p 408.

3. Kakabaev, A., Davletov, A., "A Freon Ejector Solar Cooler". Geliotekhnika, vol. 2, No. 5, pp. 42-48, 1966.

4. Barber, R. E., "Solar Air Conditioning System Using Rankine Power Cycles – Design and Test Results of Prototype Three Ton Unit". Institute of Environmental Science, 1975, pp. 170-179.

5. Miller, D. R., "Rankine Cycle Working Fluids for Solar-to-electrical Energy Conversion". Final report for Sandia Laboratory, Albuquerque, New Mexico, January, 1974.

6. Biancardi, F. R., Meader, M. D., Blecher, W. A., and Hall, J. B., "Design and Operation of Solar-powered Turbocompressor Air-conditioning and Heating System" IECEC '75 Record, pp. 186-194.

7. Allen, R. A., Stiel, L. I., "Working Fluids for Solar Rankine Heat Pumps". Solar Energy Heat Pump Systems for Heating and Cooling Buildings. ERDA Doc. coo-2560-1, 1975, pp. 98-105.

8. ASHRAE Handbook of Fundamentals, American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., New York, New York 1972.

Table 1. Some properties of the selected fluids

Selected Working Fluids	Critical Temperature (°C)	Critical Pressure (atm)	Molecular Weight (gm/gm-mole)	1 Boiler Pressure (atm)	2 Condenser Pressure (atm)	3 Evaporator Pressure (atm)	Specific Volume at Condenser Temperature (m^3/kg)	Flammable or explosive units in air (% by volume)
Steam	374	218	18.02	.78	.034	.0008	12.80	non-flammable
Refrigerant-11	198	43.2	137.38	6.97	1.10	.48	.098	non-flammable
Refrigerant-12	112	40.6	120.93	29.26	6.73	3.51	.014	non-flammable
Refrigerant-113	214	33.7	187.39	3.72	.47	.18	.208	non-flammable
Butane	152	37.46	58.13	13.25	2.53	1.20	.085	1.6 to 6.5

1 Boiler Temperature 93.3°C(200°F)
2 Condenser Temperature 26.67°C(80°F)
3 Evaporator Temperature 4.44°C(40°F)

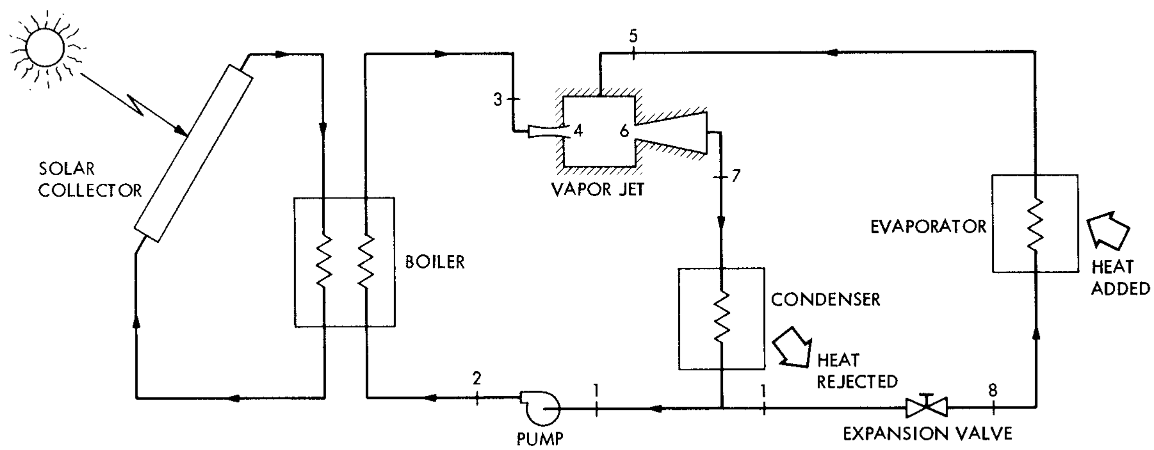

Fig. 1. Vapor jet refrigeration system driven by solar energy

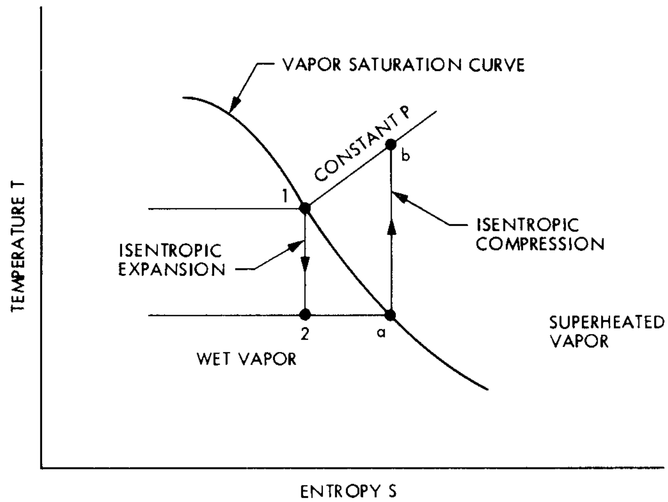

Fig. 2a. Negative slope of a vapor-saturation curve

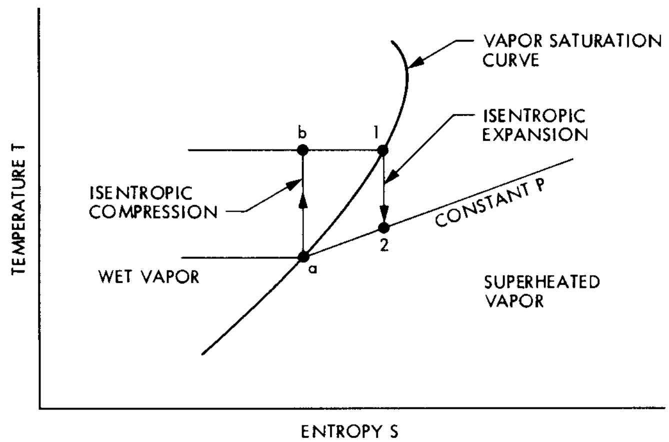

Fig. 2b. Positive slope of a vapor-saturation curve

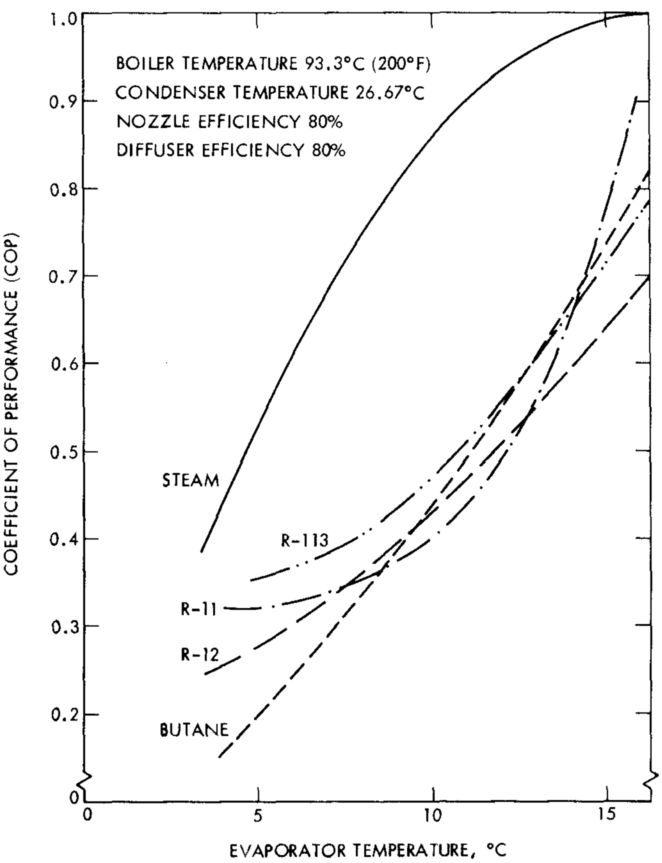

Fig. 3. Effect of the evaporation temperature on the coefficient of performance

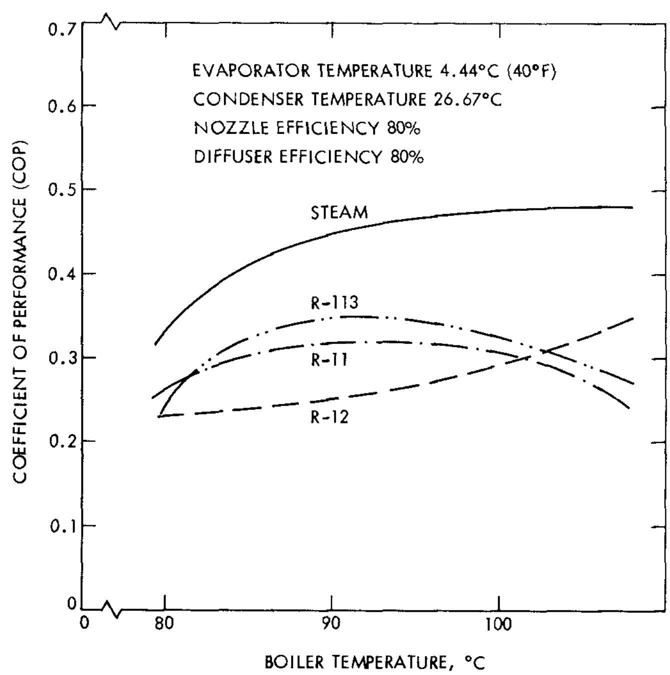

Fig. 4. Effect of the boiler temperature on the coefficient of performance